EVALUATION OF FOULING EFFECT ON THE THERMOECONOMIC OPTIMISATION OF THE GEOMETRY OF A POWER PLANT REGENERATIVE AIR HEATER

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ABSTRACT

Exergy analysis is one of the important tools that can help to discover where the available energy is inefficiently used. Therefore the purpose of this research paper is to demonstrate the importance of the use of exergy analysis in the optimisation of the geometry of a regenerative air heater and to show the effect of fouling on the optimisation objective function, which is a non-dimensionalised form of the unit cost of exergy. The optimum geometry of the regenerative air heater is determined using an objective function, derived from thermoeconomic principles. In this paper the unit cost of the exergy of the warm delivered air is used as the optimisation objective function. The unit cost is calculated from the running cost, which is determined using unit costs for the pressure component of exergy $\dot{E}^{\Delta P}$ *and the thermal component of exergy* $\dot{E}^{\Delta T}$ *which are evaluated separately. The ratio of the two unit costs that depends on the area of application, has been evaluated for a regenerative air heater used in power plant. The results show that the optimum geometry can be achieved by using exergy analysis.*

A numerical finite difference technique of heat transfer is presented for calculating the fluid and matrix temperature distributions and its effect on the regenerator performance. The governing differential equations have been formulated in terms of the characteristic dimensionless groups (Π, Λ *and* Γ).

In order to both secure high degree of accuracy of the results and to save computational time, three modifications have been made to evaluate the finite difference mesh size for regenerator length (N_r), *hot period* (N_h) *and cold period* (N_c).

Keywords: *Exergy Analysis, Thermoeconomic, Optimisation, Regenerative air heater, Fouling*

الملخص

أن التحليل الأتاحي(exergy analysis) هو من أهم الأدوات التي يمكن أن تساعد في الكشف عن الاستعمال الغير مثالي للطاقة المتاحة. بناءً عليه فإن الغرض من هذا البحث هو عرض أهمّيّة استعمال التحليل الأتاحي في العملية التفاضلية للمبادلات الحرارية الدّوارة , كذلك دراسة تأثير الاتساخ على دالة الهدف (Objective function) و التي

هي عبارة عن شكل لابعدي من أشكال تكاليف الأثاحية. الأبعاد المثالية للمبادل الحراري تم تحديـدها عـن طريـق استخدام معادلة الهدف و المشتقة من الاقتصاد الحراري. وفي هذه الدراسة تم استخدام وحدة التكلفة للهواء السـاخن الناتج كمعادلة الهدف. ومن ثم تم حساب وحدة التكلفة عن طريق تكلفة التشغيل والتي تم تقديرها باستخدام وحدة التكلفة لمركبة الأثاحية للضغط ($\dot{E}^{\Delta P}$) ولمركبة الأثاحية لدرجة الحرارة ($\dot{E}^{\Delta T}$) واللتان تم تقديرهما كلاً على حده. النسـبة بين هاتين التكلفتين تعتمد على مجال استخدام هذا النوع من المبادلات الحرارية لكن في هذا البحث تم حسابها لمحطـة توليّد الطاقة الكهربائية. لقد أظهرت النتائج أن العملية التفاضلية يمكن أن تتحقّق باستعمال التحليل الأثاحي.

تم حل المعادلات التفاضلية بعد ترتيبها بالرموز اللابعديـة (Π و Λ و Γ Dimensionless) باسـتخدام طريقـة الفارق المحدود (Finite Difference) للتحليل العددي لحساب توزيع وتأثير درجات حرارة المائع والمعدن على أداء المبادل الحراري الدوار. وحفاظاً على دقة تقديم التحليل والنتائج وكذلك وقت الكمبيوتر, فقد تم اسـتخدام التحـويرات اللازمة للحصول على حجم شبكة التحليل العددي المناسب لطول المبادل (N_r) و الوقـت الـذي أسـتغرقه المبـادل الحراري في الجانب الساخن (N_h) والجانب البارد (N_c).

NOMENCLATURE

A	Heat transfer area
A_o	Cross sectional area
c_p	Gas specific heat capacity
C_r	Heat Capacity rate (M_w c_w ω) (according to subscript)
c^ε	Unit cost of exergy
c_w	Matrix specific heat capacity
d	Matrix element hydraulic diameter
F_w	Weighting factor
$\dot{I}$	Irreversibility rate
HEE	Heat Exchanger Effectiveness
L	Regenerator matrix length
$\dot{m}$	Mass flow rate
M_w	Mass of the regenerator matrix
N_c	Finite difference cold side period intervals
N_h	Finite difference hot side period intervals
N_r	Finite difference regenerator length intervals
N_{tuo}	Number of heat transfer
Nu	Nusselt Number, h d/k
p	Period of operation (according to subscript
Pr	Prandtl Number, μ c_p/k
t	Gas temperature (according to subscript)
T_w	Regenerator matrix temperature
u	Mean velocity of fluid inside the duct
u_a	Approach velocity

Greek Symbols

γ	Ratio of principal specific heat capacities
E, ε	Exergy, specific exergy
σ	Porosity, A_o/A_{fr}
Λ	Reduced length, $hA/\dot{m}\ c_p$
μ	Fluid dynamic viscosity
Π	Reduced length, $hA/M_w\ c_w\ \omega$
ρ	Density of fluid
ρ_w	Density of matrix material
ω	Regenerator angular velocity (according to subscript
ΔP	Pressure drop
$\Delta\eta$	Steps of hot or cold period, Π /N (according to subscript)
$\Delta\xi$	Steps of regenerator length, Λ/N_r (according to subscript)

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Subscripts

ac Actual

av Average

c Cold side

e Exit

f Cold side row number in finite difference grid

fr Frontal

g Cold side column number in finite difference grid

h Hot side

i Hot side row number in finite difference grid, inlet

j Hot side column number in finite difference grid

min Minimum magnitude

max Maximum magnitude

t Total

ΔP Pressure component

ΔT Thermal component

Superscripts

ΔP Pressure component

ΔT Thermal component

1. INTRODUCTION

Energy recovery devices can have substantial impact on industrial process efficiency and their relevance to the problem of conservation of energy resources is generally recognised to be beyond dispute. One type of such a device, which is commonly used in power plant boilers, is the regenerative air heater in which a stream of hot waste gas exchanges heat with the fresh atmospheric air through the intermediate agency of a rotating matrix. As there are gas streams involved in the heat transfer process the irreversibility, or exergy destruction, due to pressure losses, $\dot{I}^{\Delta P}$ is the substantial and comparable in magnitude with that due to temperature gradients $\dot{I}^{\Delta T}$. These two principal components of total process irreversibility are not independent and there is a trade-off between them. Furthermore the cost of compensation for these forms of exergy destruction are, in general, different. It is therefore necessary when optimising the geometry of such a device to establish the relative costs of compensation applicable to the particular energy system in which the regenerator is employed. The ratio of the two unit costs $c^{\varepsilon}_{\Delta P}/c^{\varepsilon}_{\Delta T}$ expresses the relation between the two costs and is called the weighting factor, F_w. In general, F_w should be obtained for the particular energy system in which the regenerator is employed. The purpose of this research paper is to demonstrate this type of thermoeconomic technique and to show the effect of fouling on the objective function which is a non-dimensionalised form of the unit cost of exergy of the heated gas stream delivered by the regenerative air heater.

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2. Π & Λ METHOD AND EFFECTIVENESS

The classical design procedure for regenerator has been given by [Huasen , 1983]. He has considered a balanced, counterflow regenerator by using finite difference method, and obtained a relationship for the regenerator effectiveness HEE, in terms of non dimensional variables in the following form,

$$HEE = f(\Lambda,\Pi) \qquad (1)$$

where

$$\Lambda = \frac{hA}{\dot{m}\,c_p} \qquad \text{(Reduced length)} \qquad (2)$$

$$\Pi = \frac{hA\,p}{M_w c_w} \qquad \text{(Reduced period)} \qquad (3)$$

Or in terms of angular velocity ω,

$$\Pi = \frac{hA}{M_w c_w \omega} \qquad \text{(Reduced period)} \qquad (4)$$

Also [Hausen, 1983], presented the overall reduced length Λ_t and overall period Π_t where these are taken to be the harmonic means of these values, that is:

$$\frac{1}{\Lambda_t} = \frac{1}{2}\left(\frac{1}{\Lambda_h} + \frac{1}{\Lambda_c}\right) \qquad (5)$$

$$\frac{1}{\Pi_t} = \frac{1}{2}\left(\frac{1}{\Pi_h} + \frac{1}{\Pi_c}\right) \qquad (6)$$

In this paper, we are interested in determining the average fluid outlet temperatures for overall regenerator performance, thus:

$$\bar{t}_{h,e} = \frac{1}{N_h}\sum_{j=1}^{N_h} t_{h,e}(L1, j) \qquad (7)$$

$$\bar{t}_{c,e} = \frac{1}{N_c}\sum_{g=1}^{N_c} t_{c,e}(L1, g) \qquad (8)$$

where $L1=N_r+1$ (see figure 1)

In an ideal "steady-state" periodic condition, the actual heat transfer will be obtained

$$\dot{Q}_{ac} = \dot{Q}_h = (\dot{m}c_p)_h (t_{h,i} - \bar{t}_{h,e})$$

$$= \dot{Q}_c = (\dot{m}c_p)_c (\bar{t}_{c,e} - t_{c,i}) \qquad (9)$$

The maximum possible heat transfer rate, is

$$\dot{Q}_{max} = (\dot{m}c_p)_{min} (t_{h,i} - t_{c,i}) \qquad (10)$$

Thus, the regenerator effectiveness for this case is,

$$HEE_h = \frac{\dot{Q}_h}{\dot{Q}_{max}} = \frac{(\dot{m}\,c_p)_h}{(\dot{m}\,c_p)_{min}} \frac{(t_{h,i} - \bar{t}_{h,e})}{(t_{h,i} - t_{c,i})} \qquad (11)$$

$$HEE_c = \frac{\dot{Q}_c}{\dot{Q}_{max}} = \frac{(\dot{m}\,c_p)_c}{(\dot{m}\,c_p)_{min}} \frac{(\bar{t}_{c,e} - t_{c,i})}{(t_{h,i} - t_{c,i})} \qquad (12)$$

The overall regenerator effectiveness may be defined as the harmonic means of hot and cold sides effectiveness,

$$\frac{1}{HEE} = \frac{1}{2}\left(\frac{1}{HEE_h} + \frac{1}{HEE_c}\right) \qquad (13)$$

Now, since $t_{h,i}$ and $t_{c,i}$ are known as the initial conditions and $\bar{t}_{h,e}$ and $\bar{t}_{c,e}$ will be calculated using finite difference technique. Then the effectiveness as given by equations (11) and (12) can be obtained.

To facilitate the analysis and the calculations, it has been found convenient to replace Π_t with the following alternative dimensionless variables,

$$\Gamma = \frac{\dot{m}\,c_p\,p_t}{M_{w,t}\,c_w} \quad \text{For a fixed-matrix regenerator} \qquad (14)$$

$$\Gamma = \frac{\dot{m}\,c_p}{M_{w,t}\,c_w\,\omega_t} \quad \text{For a rotary regenerator} \qquad (15)$$

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where p_t is the total period and $M_{w,t}$ is the total mass of the matrix. Using the equation of continuity for a matrix we can write

$$\frac{\dot{m}}{M_{w,t}} = \frac{Re_L \mu}{\rho_w L^2 (1-\sigma)} \tag{16}$$

The matrix porosity can be defined and expressed in the following ways

$$\sigma = \frac{V_{void}}{V_{total}} = \frac{A_{\circ}}{A_{fr}} = \frac{u_a}{u} \tag{17}$$

We shell now define two Reynolds numbers. A Reynolds number based on the approach velocity (outside the matrix) and the matrix length L.

$$Re_L = \frac{\rho u_a L}{\mu} \tag{18}$$

And the duct (matrix passage) Reynolds number is,

$$Re = \frac{\rho u d}{\mu} \tag{19}$$

Using expressions (14),(15) and (18) we can express Γ in the following forms.

$$\Gamma = \frac{c_p}{c_w} \frac{\mu Re_L}{\rho_w L^2} \frac{p_t}{(1-\sigma)} \quad \text{For fixed-matrix regenerator} \tag{20}$$

$$\Gamma = \frac{c_p}{c_w} \frac{\mu Re_L}{\rho_w L^2} \frac{1}{(1-\sigma)\omega_t} \quad \text{For a rotary regenerator} \tag{21}$$

Similarly, Λ may be written as follows for a square duct

$$\Lambda = \frac{4L^2}{\mu c_p Re_L} \frac{h\sigma}{d} \tag{22}$$

Expressions (20),(21) and (22) show more explicitly then (14), (15) and (2) which are the basic variables in the physical phenomenon under study.

3. PRESSURE DROP AND HEAT TRANSFER

Since a fully developed flow is seldom realised in a regenerator matrix, expressions for laminar developing flow (hydrodynamically and thermally) are used in the present study. [Shah, 1978] provided a correlation for the apparent fanning friction factor f_{app} in the following form:

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$$f_{\text{app}}\,\text{Re} = 3.44(x^{+})^{-0.5} + \frac{\dfrac{K(\infty)}{4x^{+}} + f\,\text{Re} - 3.44(x^{+})^{-0.5}}{1 + \text{C}\,(x^{+})^{-2}} \tag{23}$$

where $x^{+} = \dfrac{\text{L}}{\text{d Re}}$ (24)

Here d is the hydraulic diameter and $K(\infty)$ is the incremental pressure drop number which has a constant value in the fully developed flow. f stands for the fanning friction factor for the fully developed region and C is a constant coefficient.

f_{app} takes into account both the skin friction and the change in momentum rate in the hydrodynamic entrance region. It is based on the static pressure drop from x = 0 to L. It is defined by

$$f_{app}\,\frac{4\text{L}}{d} = \frac{\Delta\text{P}}{\rho u^{2}/2} \tag{25}$$

If we substitute equation 25 into 23 and rearrange, we have

$$\Delta\text{P} = \frac{\rho u^{2}\,2\text{L}}{d\ \text{Re}} \times \left\{ 3.44(x^{+})^{-0.5} + \frac{\dfrac{K(\infty)}{4x^{+}} + f\,\text{Re} - 3.44(x^{+})^{-0.5}}{1 + \text{C}\,(x^{+})^{-2}} \right\} \tag{26}$$

$K(\infty)$, f_{app} and C of equation 26 for square ducts are presented in Table 1.

Table 1. Square duct parameters for equation 23

$K(\infty)$	f Re	C
1.43	14.227	0.00029

[Chandrupatla and sartri, 1977] solved the thermal entrance length problem by neglecting the effects of viscous dissipation, fluid axial conduction, and thermal energy sources in the fluid. The results of their calculations applicable to a Newtonian fluid for the case when the wall temperature is assumed constant around the perimeter at any given cross section have been expressed in form of a polynomial as follows

$$Nu = 3.612 + 0.0831\left(\frac{1}{x^{*}}\right) - 0.0004131\left(\frac{1}{x^{*}}\right)^{2} \tag{27}$$

where x is a non- dimensional group defined by

$$x^* = \frac{L}{d\ \mathrm{Re}\,\mathrm{Pr}} \tag{28}$$

Colburn Factor J is defined as

$$J = St\ Pr^{2/3} = (Nu\ Pr^{-1/3}) / Re \tag{29}$$

But Stanton number, St , is given by

$$St = (h/\rho\ u\ c_p) \tag{30}$$

Combining equations 29 and 30, we have

$$h = (Nu/\ Pr\ Re)\ (\rho\ u\ c_p) \tag{31}$$

The heat transfer coefficient h is used to find the non-dimensional groups N_{tuo} and Λ (equation 22)

4. THE INFLUENCE OF FOULING UPON PRESSURE DROP AND HEAT TRANSFER

In the design of the regenerators, the enhancement of heat transfer surface area is effective to reduce the loss due to the fluid-to fluid temperature difference. It, however, leads to the increase of the pressure loss in the passages. Pressure drop is an important factor in the exergy method of optimisation of the geometry of regenerators. Its importance relative to the heat transfer effect depends on the regenerator and heat transfer loading. Therefore the relationships for thermal and hydrodynamic performance can be expressed as follows

$$\text{Colburn Number}\quad J = C\ Re^{-n} \tag{32}$$

$$\text{Friction Factor}\quad f = D\ Re^{-p} \tag{33}$$

where C, n, D, and p are matrix element performance characteristics and presented as follows [National Power-Technology and Environmental Centre, U.K.],

$C = 0.3192$, $n = 0.5900$, $D = 1.8500$, and $p = 0.6620$

Fouling refers to the undesired accumulation of solid material on regenerator surface which increases thermal resistance to heat transfer, and may also increase the pressure drop. [Shah 1985] has presented a study of the effect of fouling in the compact heat exchanger. He pointed out that, the friction factor f is increased by 15 to 30 percent, while the Colburn factor J is decreased by 5 to 10 percent. In this research paper, the maximum effect for fouling will be considered. Hence equations 32 and 33 become.

Colburn Number $J = 0.9\ C\ Re^{-n}$ (34)

Friction Factor $f = 1.3\ D\ Re^{-p}$ (35)

5. FINITE DIFFERENCE SOLUTION

Figure (1) represents schematically a typical element from each fluid side on the regenerator. And Figure(2) shows a typical element motion. Now consider the energy balance for an element in which the energy transfer to the element by convection must equal the energy storage in the element. Then the relation over a differential element of the regenerator may be expressed as only for the regenerator elements on side of Λ_h [Jassim, 1992].

$$\dot{Q}_h = \frac{(hA)_h}{\Lambda_h}[t_h(i,j) - t_h(i+1,j)](1/N_h) \tag{36}$$

$$\dot{Q}_h = (hA)_h \Delta T_{av,h}(1/N_r N_h) \tag{37}$$

$$\dot{Q}_h = \frac{(hA)_h}{\Pi_h}[T_{w,h}(i,j+1) - T_{w,h}(i,j)](1/N_r) \tag{38}$$

And $\Delta T_{av,h}$ represent the mean temperature difference between the gas and the matrix element. For small enough element the arithmetic mean temperature difference may be assumed valid so that,

$$\Delta T_{av,h} = (1/2)[t_h(i,j) + t_h(i+1,j)] - (1/2)[T_{w,h}(i,j) + T_{w,h}(i,j+1)] \tag{39}$$

By using equations (36), (37), (38) and (39), the outlet air and matrix temperatures for an element on the side of Λ_h can be obtained as follows (see appendix A)

$$t_h(i+1,j) = t_h(i,j) - B1[t_h(i,j) - T_{w,h}(i,j)] \tag{40}$$

$$T_{w,h}(i,j+1) = T_{w,h}(i,j) + B2[t_h(i,j) - T_{w,h}(i,j)] \tag{41}$$

Similarly, for an element on the side of Λ_c the heat transfer rate is

$$\dot{Q}_c = \frac{(hA)_c}{\Lambda_c}[t_c(f+1,g) - t_c(f,g)](1/N_c) \tag{42}$$

$$\dot{Q}_c = (hA)_c \Delta T_{av,c}(1/N_r N_c) \tag{43}$$

$$\dot{Q}_c = \frac{(hA)_c}{\Pi_c}[T_{w,c}(f,g) - T_{w,c}(f,g+1)](1/N_r) \tag{44}$$

and

$$\Delta T_{av,c} = (1/2)[T_{w,c}(f,g) + T_{w,c}(f,g+1)] - (1/2)[t_c(f,g) + t_c(f+1,g)] \tag{45}$$

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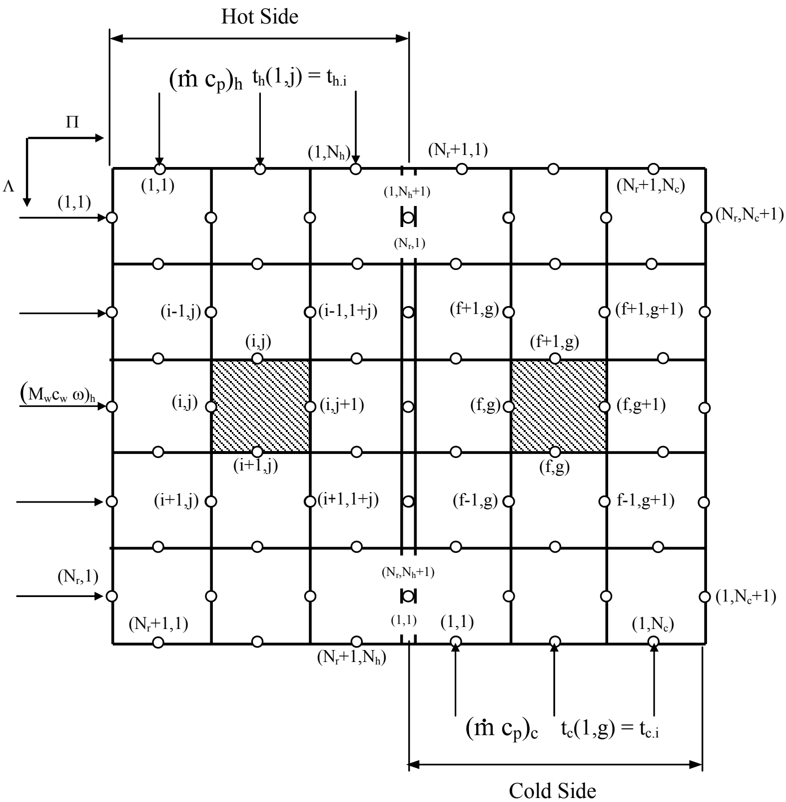

Figure 1. Schematic representation of a rotary regenerator

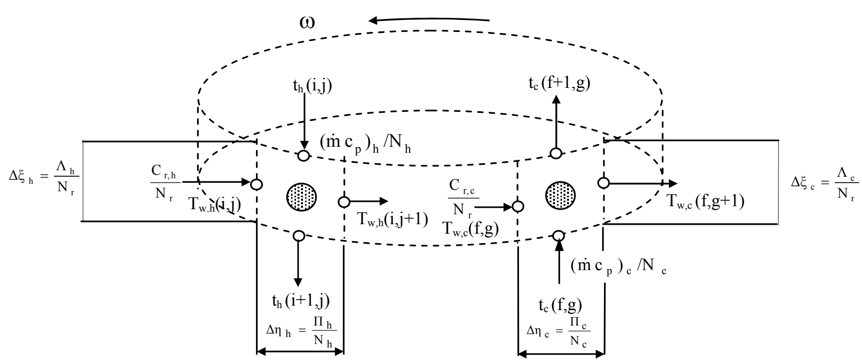

Figure 2. A typical element motion (i,j) on the side of Λ_h and (f,g) on the side of Λ_c

By the same method, the outlet temperatures of the elements on the side of Λ_c may be obtained by using equations (42), (43), (44) and (45)

$$t_c(f+1,g) = t_c(f,g) + B3\,[T_{w,c}(f,g) - t_c(f,g)] \tag{46}$$

$$T_{w,c}(f,g+1) = T_{w,c}(f,g) - B4\,[T_{w,c}(f,g) - t_c(f,g)] \tag{47}$$

By using equations (40), (41), (46) and (47) , the outlet fluid and matrix temperatures for the elements on the side of Λ_h and Λ_c , can be obtained (see appendix A).

6. CHOICE OF N_r, N_h and N_c

If the total dimensionless reduced length of the regenerator was Λ' [van-Leersum, 1986], a numerical experiment show that the finite difference equation will have stable and convergent solution if,

$$N_r = 1.7\sqrt{\Lambda'} + 3.2 \tag{48}$$

where Λ' modified dimensionless length for the effect of condensation.

In this paper Λ' have been modified to Λ_t (see equation 5) and found, if the number 3.2 is doubled the results will be in a good agreement with [Hausen, 1983] results. Then

$$N_r = 1.7\sqrt{\Lambda_t} + 6.4 \tag{49}$$

And in order to find N_h and N_c, equation (48) is also modified to,

$$N_h = 1.7\sqrt{\Pi_t} + 6.4 \tag{50}$$

$$N_c = 1.7\sqrt{\Pi_t} + 6.4 \tag{51}$$

7. CONVERGENCE CRITERIA

The Convergence criteria which have been used are, the steady state criteria to insure that steady cyclic conditions are established, the condition for a balanced regenerator is,

$$\left|\left(\frac{\Lambda}{\Pi}\right)_h - \left(\frac{\Lambda}{\Pi}\right)_c\right| \le \delta 1 \tag{52}$$

$\delta 1$ should be equal to zero in the balanced regenerator, $\delta 1$ is zero if the physical properties of the air and matrix in both sides are evaluated at the average temperature for the inlet fluids,

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but if the physical properties of air and matrix are evaluated at the average temperature for the inlet and outlet for each side, therefore it will be impossible for δ1to be zero. In this work the physical properties have been evaluated at the average temperature for the inlet and outlet for each side. Hence the convenient value for the convergence criteria is found to be 0.00025. Under this condition, it is often convenient to determine weather dynamic equilibrium has been reached.

In addition to equation (52), two alternative additional conditions are required. These are

1- If in the first condition equation (52), the RHS is less than δ1 (balanced regenerator), then to reach the steady state we have to have

$$\left|HEE_h - HEE_c\right| \le \delta 2 \tag{53}$$

where δ1 = 0.0025

If equation (53) is satisfied, then the result will be printed out. Otherwise, the left edge of the regenerator is physically the same as the right edge, therefore, the matrix temperatures for the elements of the first column on the side of Λ_h are the same as the outlet temperatures for corresponding elements of the last column on the side Λ_c of . Then

$$T_{w,h}(i, j) = T_{w,c}(M3, N4) \tag{54}$$

where

$$i = 1,2, \rightarrow N_r$$
$$M3 = (N_r + 1 - i)$$
$$N4 = N_c + 1$$

2- If however the condition equation is not satisfied, the regenerator is unbalanced. Under these conditions one of the following must be satisfied.

$$\left|(HEE_h)_i - (HEE_h)_{i-1}\right| \le \delta 2 \tag{55}$$

$$\left|(HEE_c)_f - (HEE_c)_{f-1}\right| \le \delta 2 \tag{56}$$

If either equation (55) or equation (56) is satisfied, the result will be print out. Otherwise the program will return to equation (54) and repeat the calculation.

8. OPTIMISATION OBJECTIVE FUNCTION

It is common practice in thermoeconomic to select the unit cost of the product c^{ε}_{PROD} , as the optimisation objective function. The unit cost of the product will be obtained by dividing the cost rate of operating of the regenerator by the output. The cost rate of operation consists principally of the cost of investment and the running cost. Because of lack of cost data, the cost of investment is excluded. Consequently the unit cost of the product will be expressed as:

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$$c^{\varepsilon}_{PROD} = \frac{\dot{C}_R}{\dot{E}_{out}} \tag{57}$$

where $\dot{C}_R$ is the running cost rate and $\dot{E}_{out}$ is the exergy flow rate of the heated air.

The exergy balance for the regenerator can be expressed as [Kotas, and Jassim, 1993],

$$\dot{E}^{\Delta T}_{h,i} - \dot{E}^{\Delta T}_{h,e} + \dot{E}^{\Delta P}_{h,i} - \dot{E}^{\Delta P}_{h,e} - \left[\dot{E}^{\Delta T}_{c,e} - \dot{E}^{\Delta T}_{c,i} - \left(\dot{E}^{\Delta P}_{c,i} - \dot{E}^{\Delta P}_{c,e}\right)\right] = \dot{I} \tag{58}$$

In this expression each exergy flux is split into the pressure component $\dot{E}^{\Delta P}$ and the thermal component $\dot{E}^{\Delta T}$. These are, respectively, zero when the stream pressure $P = P_o$ or temperature $T=T_o$. In this paper, P_o is considered as the exhaust pressure of both streams and T_o as the inlet temperature of the cold.

One of the LHS of equation (58) can now be identified as desired output and those remaining as the input. These are:

$$\dot{E}_{out} = \dot{E}^{\Delta T}_{c,e} + \dot{E}^{\Delta T}_{h,e}$$

$\dot{E}^{\Delta T}_{h,e}$ is released to the atmosphere and its not beneficial in the present analysis, therefore is neglected.

$$\dot{E}_{out} = \dot{E}^{\Delta T}_{c,e} \tag{59}$$

and

$$\dot{E}_{in} = \dot{E}^{\Delta T}_{h,i} + \dot{E}^{\Delta P}_{h,i} + \dot{E}^{\Delta P}_{c,i} \tag{60}$$

The idealisation of the input and output terms carried out above is based on the same principles as those used in the formulation of the exergetic efficiency of a heat exchanger [Kotas et al, 1991].

Since the input consists of both forms of exergy, $\dot{E}^{\Delta T}$ and $\dot{E}^{\Delta P}$, they will be allocated appropriate unit costs $c^{\varepsilon}_{\Delta T}$ and $c^{\varepsilon}_{\Delta P}$ respectively. Hence, we get the cost rate of the input in the following form:

$$\dot{C}_{in} = c^{\varepsilon}_{\Delta T}\,\dot{E}^{\Delta T}_{h,i} + c^{\varepsilon}_{\Delta P}\left[\dot{E}^{\Delta P}_{h,i} + \dot{E}^{\Delta P}_{c,i}\right] \tag{61}$$

If the running cost, $\dot{C}_R$ is regarded as being identical with $\dot{C}_{in}$, the objective function may be written in a dimensionless form as a cost ratio.

$$R_c = \frac{c^{\varepsilon}_{PROD}}{c^{\varepsilon}_{\Delta T}} = \frac{\dot{E}^{\Delta T}_{h,i} + F_w\left[\dot{E}^{\Delta P}_{h,i} + \dot{E}^{\Delta P}_{c,i}\right]}{\dot{E}^{\Delta T}_{c,e}} \tag{62}$$

where

$$F_w = \frac{c^{\varepsilon}_{\Delta P}}{c^{\varepsilon}_{\Delta T}} \tag{63}$$

Introducing the physical exergy relations of a perfect gas into equation (62), we get

$$R_c = \frac{t_{h,i} - T_o - T_o\, ln\left[\frac{t_{h,i}}{T_o}\right] + F_w\left[\frac{\gamma - 1}{\gamma}\right] T_o \left[ln\left(1 + \frac{\Delta P_h}{P_o}\right) + ln\left(1 + \frac{\Delta P_c}{P_o}\right)\right]}{\bar{t}_{c,e} - T_o - T_o\, ln\left[\frac{\bar{t}_{c,e}}{T_o}\right]} \tag{64}$$

where $\Delta P_h = P_{h,i} - P_o$ and $\Delta P_c = P_{c,i} - P_o$

Note that the exit temperature $\bar{t}_{c,e}$ is an average temperature, owing to the fact that the matrix temperature varies with angular coordinate.. To evaluate R_c, it is necessary to obtain the value of $\bar{t}_{c,e}$ and the pressure drops ΔP_h and ΔP_c. These quantities are evaluated with the aid of a computer program EXERGEN [Jassim, 1992] using a finite difference technique.

9. WEIGHTING FACTOR

As defined by equation (63), the weighting factor is a ratio of the unit cost of the pressure component of exergy $c^{\varepsilon}_{\Delta P}$, to that of the thermal component of exergy, $c^{\varepsilon}_{\Delta T}$. The weighting factor is found to be 7.64 by [Kotas and Jassim 1993] for an electric power plant in which the rotary regenerator is used.

10. NUMERICAL EXAMPLE

The optimisation of the geometry L/d of the passages in the matrix and a study of the evaluation of the effect of fouling has been carried out on a regenerative air heater used in power station having the following parameters,

Mass flow rate of air	$\dot{m}_h = \dot{m}_c = 337$ kg / s
Gas inlet temperature	$t_{h,i} = 609.15$ K
Air inlet temperature	$t_{c,i} = 305.15$ K
Environmental temperature	$T_o = 273.15$ K
Environmental pressure	$P_o = 1$ bar
Regenerator diameter	14.61 m
Rotor area	$A_{fr} = 141.7$ m^2
Regenerator length	1.931 m
Total Period (p_t)	60 s
Matrix porosity	0.8
Weighting factor	7.64
Material	Mild Steel

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11. RESULTS AND DISCUSSION

Figures 3 and 4 show the effect of the ratio L/d on the performance criteria R_c and HHE. In terms of the objective function, R_c, the optimum hydraulic diameter is 3.1 mm and the corresponding values of heat exchange effectiveness is 0.93. The sensitivity of the criteria of performance, to changes in the ratio L/d is examined by considering the effect of using values of the ratio which correspond to values of R_c 10% higher than the minimum. The result are shown in Table 2. From this table a suitable value of hydraulic diameter, d, can be selected to give gas outlet temperature, $\bar{t}_{h,e}$, high enough to ensure adequate gas plume buoyancy.

Table 2 Performance criteria and HEE results without considering the effect of fouling

L/d	d/mm	R_c	HEE	$\bar{t}_{h,e}$ K
350	5.5	1.386	0.83	356.85
623	3.1	1.26	0.93	326.45
1135	1.7	1.386	0.97	314.25

Figures 3 and 4 also show the effect of fouling on the criteria of performance R_c and HEE. This has been obtained using [Shah 1985] estimates of the effect of fouling for compact heat exchangers. The results are summarised for both the minimum value of R_c and that 10% above minimum in Table 3.

Table 3 Performance criteria and ε results with considering the effect of fouling

L/d	d/mm	R_c	HEE	$\bar{t}_{h,e}$ K
350	5.5	1.441	0.80	365.95
570	3.4	1.31	0.92	330.65
965	2.0	1.441	0.96	317.35

As will be seen by comparing these results with those in Table 2, the effect of fouling appears to be less than is generally thought to be the case. However, it must be noted that Shah's estimate of the effect of fouling on the friction factor and the heat transfer coefficient is based on the form of fouling found in compact heat exchangers which may not be typical of the type of deposits found on fouled matrix elements is coal fired power plants.

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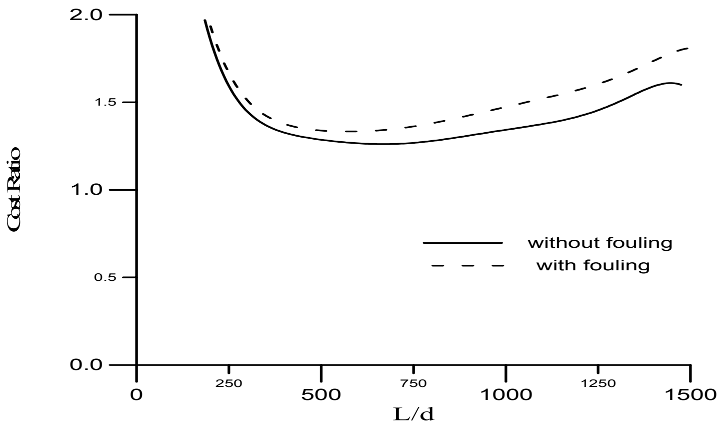

Figure 3 Effect of Fouling on the cost ratio R_c and L/d

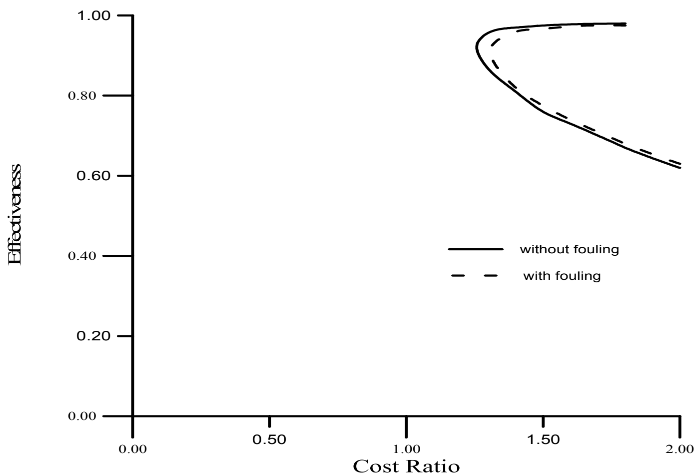

Figure 4 Effect of Fouling on the cost ratio R_c and the effectiveness HEE

12. CONCLUSIONS

As shown in the above example, the cost ratio is made up of the costs of compensation for the destruction of the two components of exergy $\dot{E}^{\Delta T}$ and $\dot{E}^{\Delta P}$. Since the compensation for exergy destroyed in the heat transfer process has to come from the earth's natural resources, the objective function used in this paper may be directly relevant to the problem of conservation of natural resources.

The investigation of the effect of fouling, whilst providing some insight into the possible consequences, suffers from the fact that the information on the effect of fouling on the friction factor and the heat transfer coefficient has been taken from another area of heat transfer technology. It would therefore be advisable to obtain information on the effect of fouling for the actual conditions prevailing in the matrix of a regenerative air heater.

One of the features of the optimisation technique presented in this work is the use of the weighting factor, F_w, which reflects the different values of the unit costs of the two components of exergy. Although in the present work the value of F_w was determined from thermoeconomic considerations of the exergy conservation processes in the plant in which the regenerator is used. Clearly the use of the weighting factor can be extended to other types of heat exchangers.

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APPENDIX A

A- When $C_c = (\dot{m}\ c_p)_c = C_{min}$

$$\Lambda_h = C^*\left[1+\frac{1}{(hA)^*}\right]N_{tuo} \quad , \quad \Pi_h = \frac{1}{C_R^*}\left[1+\frac{1}{(hA)^*}\right]N_{tuo}$$

where

$$C^* = \frac{(\dot{m}c_p)_{min}}{(\dot{m}c_p)_{max}} \quad , \quad C_r^* = \frac{M_w c_w\ \omega}{(\dot{m}c_p)_{min}}$$

$$N_{tuo} = \frac{1}{C_{min}}\left[\frac{1}{\frac{1}{(hA)_c}+\frac{1}{(hA)_h}}\right] \quad , \quad (hA)^* = \frac{(hA)_{min}}{(hA)_{max}}$$

and

$$B1 = \frac{2}{\frac{2N_r(hA)^*}{C^*\left[(hA)^*+1\right]N_{tuo}}+\frac{N_r}{N_h C^* C_r^*}+1} \quad , \quad B2 = \frac{2}{\frac{2\ C_r^* N_h (hA)^*}{\left[(hA)^*+1\right]N_{tuo}}+\frac{N_h C^* C_r^*}{N_r}+1}$$

$$B3 = \frac{2}{\frac{2N_r}{\left[(hA)^*+1\right]N_{tuo}}+\frac{N_r}{N_c C_r^*}+1} \quad , \quad B4 = \frac{2}{\frac{2N_r C_r^*}{\left[(hA)^*+1\right]N_{tuo}}+\frac{N_c C_r^*}{N_r}+1}$$

B- When $C_h = (\dot{m}\ c_p)_h = C_{min}$, the values of B1, B2, B3 and B4 will be

$$B1 = \frac{2}{\frac{2N_r}{\left[(hA)^*+1\right]N_{tuo}}+\frac{N_r}{N_h C_r^*}+1} \quad , \quad B2 = \frac{2}{\frac{2\ N_r C_r^*}{\left[(hA)^*+1\right]N_{tuo}}+\frac{N_h{}^* C_r^*}{N_r}+1}$$

$$B3 = \frac{2}{\frac{2N_r(hA)^*}{C^*\left[(hA)^*+1\right]N_{tuo}}+\frac{N_r}{N_c C^* C_r^*}+1} \quad , \quad B4 = \frac{2}{\frac{2N_c C_r^*(hA)^*}{\left[(hA)^*+1\right]N_{tuo}}+\frac{N_c C^* C_r^*}{N_r}+1}$$

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